Design and stability of a floating free stream energy converter

W.M.J. Batten[1] , F. Weichbrodt[2], G.U. Muller[1],J. Hadler[3], C. Semlow[3],
M. Hochbaum[2], S. Dimke[2], , P. Fröhle[2]
[1]School of Civil Engineering & the Environment,
University of Southampton,
Southampton, SO17 1BJ, UK
[2] Coastal Engineering Research Group, Inst. of Environmental Engineering,
[3] Faculty of Mechanical Engineering and Marine Technology,
University of Rostock
18059 Rostock, Albert-Einstein-Strasse, Germany
E-mail: wmjb@soton.ac.uk

Abstract: *A floating energy converter to extract kinetic energy from river and sheltered tidal sites has been developed and tested. The energy converter has a floating body with side pontoons, base plate and separators that drives a hydrostatic pressure wheel. This paper compares results from floating lab scale models 1.2m in length with 0.5m diameter wheel tested in a circulating water channel with a 7.6m long demonstrator towed in an estuary. The power and stability scaling of the models are assessed and the results are used to predict performance of larger devices to assess potential in remote areas.*

Keywords: *Free Stream Energy Converter, Boat Mill.*

1. INTRODUCTION

Boat mills were common site on many large rivers throughout Europe from medieval times to 20th century and were generally used for grinding food sources (Gräf, 2006). In general the water wheels had diameters of around 5-6 m and widths of a similar size. Most mills had a main hull to house the gearing and milling equipment and a side pontoon to stabilize the wheel. Typically the main hulls had length of 12-16 m with a width of 4-5 m. The side pontoons were around a third the width of the main hull. The key advantage of boat mill was that they could be easily installed on fast flowing sections of large rivers, such as between bridges and on bends. However, powers produced were significantly less than that of fixed water wheels installations working on head differences (Müller & Kauppert, 2004; Müller & Batten, 2010).

Today there is a new requirement to extract energy from renewable sources such as river and tidal currents. The current work developed is based upon improving the classic designs, which separated the design of the wheel from the hull forms, to a design that uses the hull form to improve the performance of the wheel. Previous investigations (Page *et al.*, 2009; Batten & Müller, 2011) have demonstrating improved efficiency of energy extraction by using hull form shape, a base plate and separators to increase the energy flow through the wheel. The aim of the work presented is to demonstrate floating stability and scaling from lab geometry to a large scale model.

The work also forms a part of a wider FP7 energy project “Hydropower converters for very low head differences” (Müller, 2009; HYLOW, 2010).

2. SMALL SCALE MODEL

2.1. Outline

Based upon the previous work a generic geometry form has been developed; the hull form and layout is shown in the schematic in Figure 1. The bow section has a contraction region which is designed for maintain a constant flow velocity and a constriction which allows the development of a head in front of the turbine. The stern section also has an expansion section that is designed for the flow to exit at a shallower depth with a higher velocity. The scoop at the front acts to enhance the inlet constriction whilst the separators are designed to generate a region of low pressure behind the model and further reduce the water level behind the rotor. The developed head difference causes a moment to be generated which must be counter acted by ballast to maintain a correct trim angle.

In order for the design to also work in tidal estuaries it is preferable that the design is symmetrical so a large turning circle is not required. For the design to be cost effective choosing a hull form that has minimum length desirable due to material and fabrication costs. The scoop and separators could be hinged to alter position as the tide changes in order to make a bi-directional unit and could be used to control the maximum power.

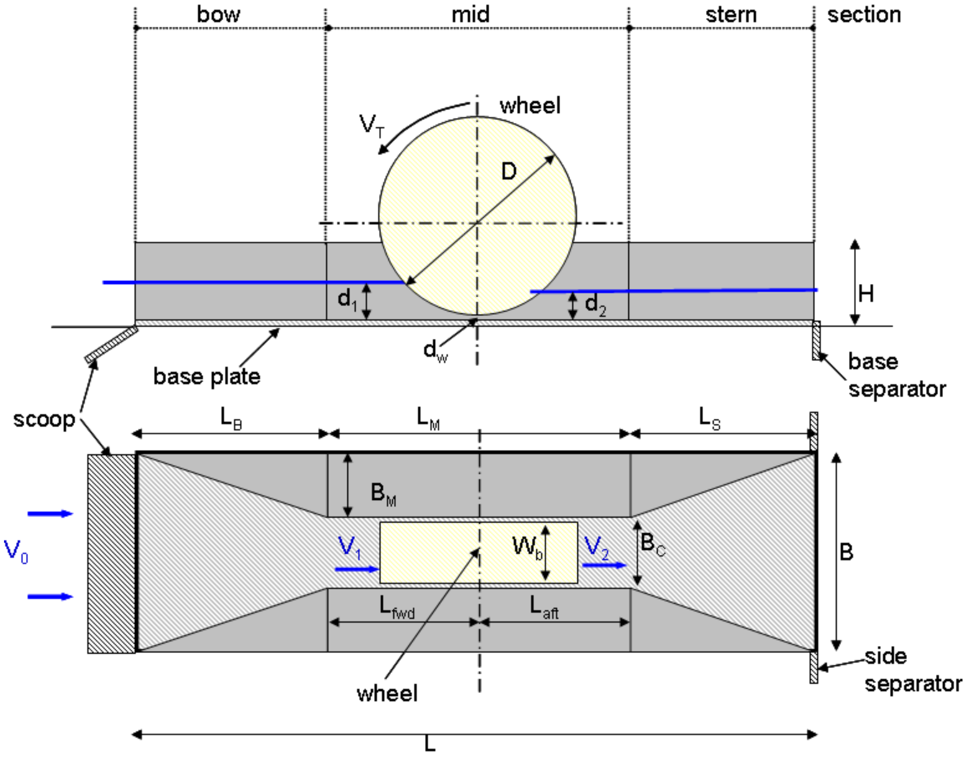


Figure 1 General schematic showing principle dimensions

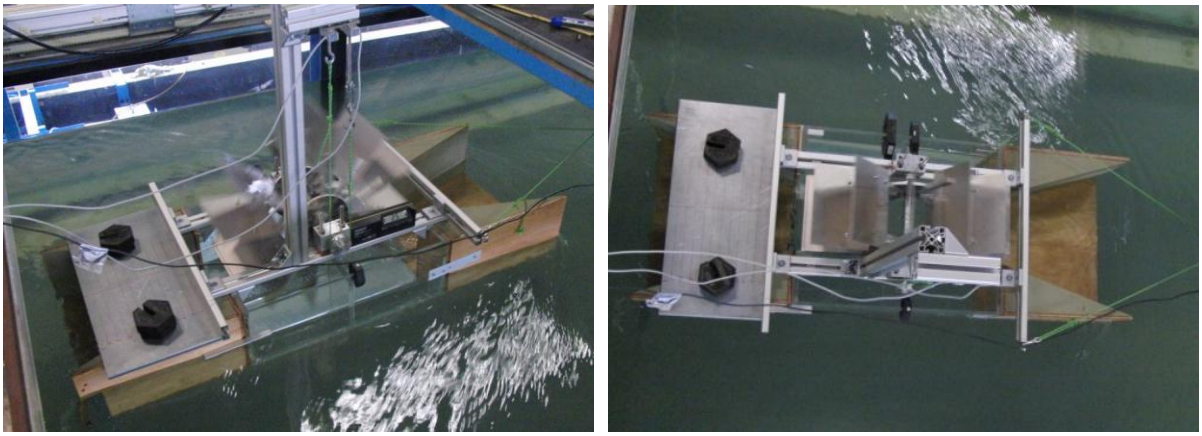
Table 1 Principle model and test dimensions

Scale	**Value**	**Comment**
Speed (V_0)	0.40 m/s	The speed was held constant during each set of tests
Beam (B)	420 mm	
Wheel breadth (W_b)	176 mm	Tolerance +/- 0.2 mm
Length (L)	1220 mm	
Midsection length (L_M)	500 mm	
Wheel diameter (D)	500 mm	
Wheel clearance (d_w)	16 mm	
Draft	100mm	External draft was set constant for all tests

2.2. Test facility, model and setup

The tests were performed in the circulating water channel at hydraulics laboratory of Technische Universität Braunschweig, Germany. The 30m tilting flume has a width of 2m and a maximum draft of 800mm. The test models were fixed rigidly in the tank about 20m from the inlet in the centre. The flume was tilted (s = 0.0005) to ensure uniform flow conditions. The free stream velocity upstream of the model was measured by a Sontek ADV system. This was placed a model width upstream and velocities were recorded with both the model wheel stationary and rotating.

Details of the model geometry are shown in Figure 1 and the principle dimensions are presented in Table 1. The mid section is made of Acrylic to view the internal water depths. The bow and stern sections are made from foam and wood with 26.6° inflow angles. Figure 2 shows a photograph of the model centred in the test tank floating from a yoke mooring which was attached to a load cell. The wheel was mounted on stainless steel ball bearings with no seals to ensure low friction. The power take-off consisted of a *Prony*-brake. The tests were performed at nominal speed of 0.4 m/s at a constant draft of 100 mm. The model was trimmed manually by moving weights the moment was recorded. Further details of the model setup and static results are discussed in Batten & Müller (2011).



(a) Floating model showing bow up trim angle **(b) Floating model viewed from above**

Figure 2 Setup in test tank with the wheel under load with 90° separators; Flume depth 600 mm and flow speed 0.40 m/s

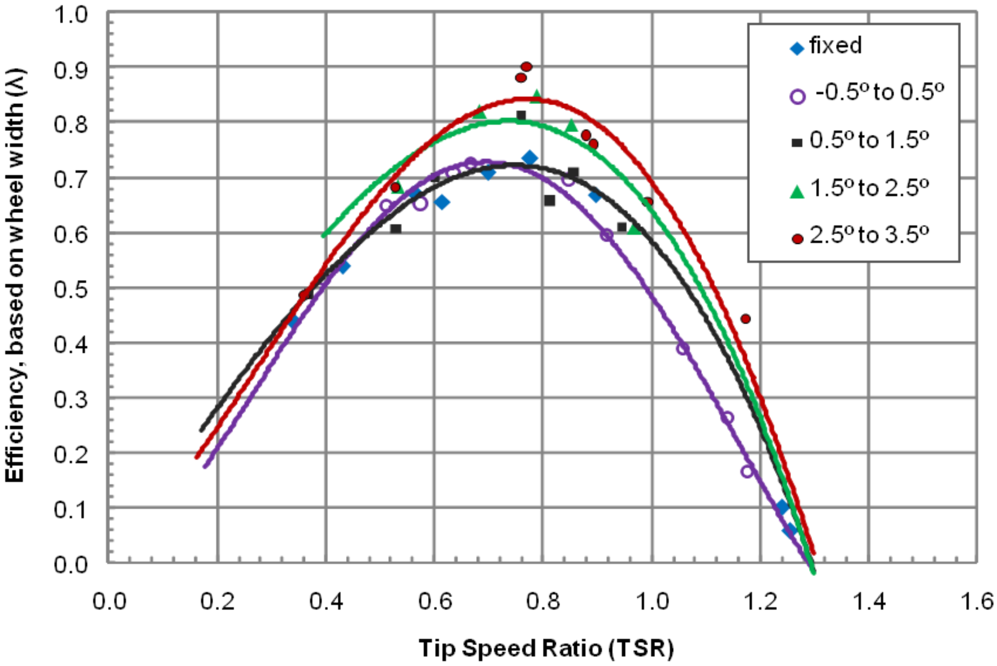


Figure 3 Effect of trim angle on floating model performance (negative angle bow down)

2.3. Data presentation and results

The results are presented in non-dimensional form to allow comparisons and predictions for full scale devices. The blade speed is presented as a blade tip ratio;

$$TSR = \frac{V_T}{V_0} \tag{1}$$

where V_T is the tip speed of the blade and V_0 is the free stream speed measured from the ADV as discussed in section 3.1. The efficiency of the device is calculated from the hydrodynamic kinetic energy based on blade area and the wheel shaft power and is defined as;

$$\lambda = \frac{P_s}{0.5\rho\, A\, V_0^3} \tag{2}$$

where P_S is the shaft power, ρ is the water density and A is the blade area.

The floating model results for a range of trim angles is compared to fixed model tests presented in Batten & Müller (2011). Although there is some scatter in the results due to experimental accuracy the results show that the fixed model performed slightly better than the level model (-0.5° to +0.5°). This is due to the slight motions of the model. The results also clearly show that trimming the model bow down significant performance gains can be achieved as the peak efficiency has increased by over 20%. The mechanism for this performance gain is still being investigated but larger head difference was evident and possibly less water was lifted by the wheel as the blade exited the water.

3. LARGE SCALE MODELS

3.1. Design and construction

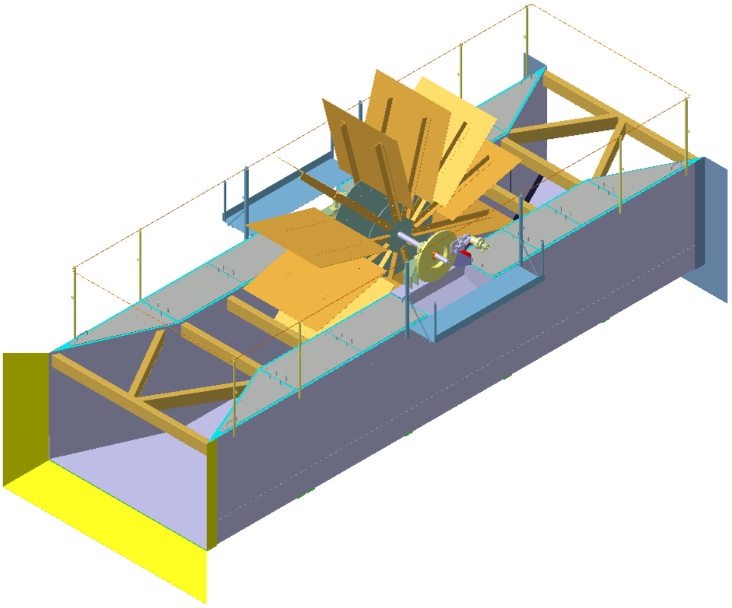
The length and width of the large scale model were determined by transport cost considerations. The key dimensions of the model are in Table 2 scaled and inferred from small and medium scale tests. The large scale model is constructed from steel and the hull consists of two pontoons connected by a base plate. Each pontoon has a series of water tight bulkheads for ballast and trimming the model.

The wheel has twelve steel plate blades which are contacted to a shaft via spokes and a collar. Figure 4(a) shows a CAD rendering of the model that also detailing the stiffening beams on the top between the pontoons. The power is dissipated using a disk brake to simulate power takeoff (Figure 4(b)). The callipers for the disk brake are mounted on load cells which measure the reaction force. The power is then calculated from the calibrated load cells and shaft encoder.

The floating model is shown in Figure 4(c). Further details of the design are presented in Hadler & Broekel (2010).

Table 2 Principle dimensions of the large scale model

Scale	Value	Comment
Speed (V_0)	0.5 - 1.5 m/s	Towed by an electro-hauler
Beam (B)	2.4 m	-
Length (L)	7.6 m	Excluding scoop and separator
Wheel breadth (W_b)	1.14 m	-
Wheel diameter (D)	3.2 m	-
Draft	0.4 - 0.75 m	Draft was varied to access performance
Wheel clearance (d_w)	0.05 - 0.15 m	The gap at the base was varied to assess sensitivity



(a) CAD rendering of the large scale model (Hadler & Broekel, 2010 & 2010)

(b) Disk break and callipers instrumented with load-cell

(c) Model setup and ready for towing tests

Figure 4 Details of the large scale model

3.2. Stability

The total weight of the large scale model without any ballast is estimated with 4,300 kg. Based on dimensions this produces an estimated minimal draught of the large scale model is 473 mm. For operation drafts up to 1000mm are required. The pontoons are deigned to be filled with either sand or water ballast. Water ballast allows the model. The additional ballast material for draughts of 0.75 m and 1 m has to be 2,520 kg and 4,800 kg.

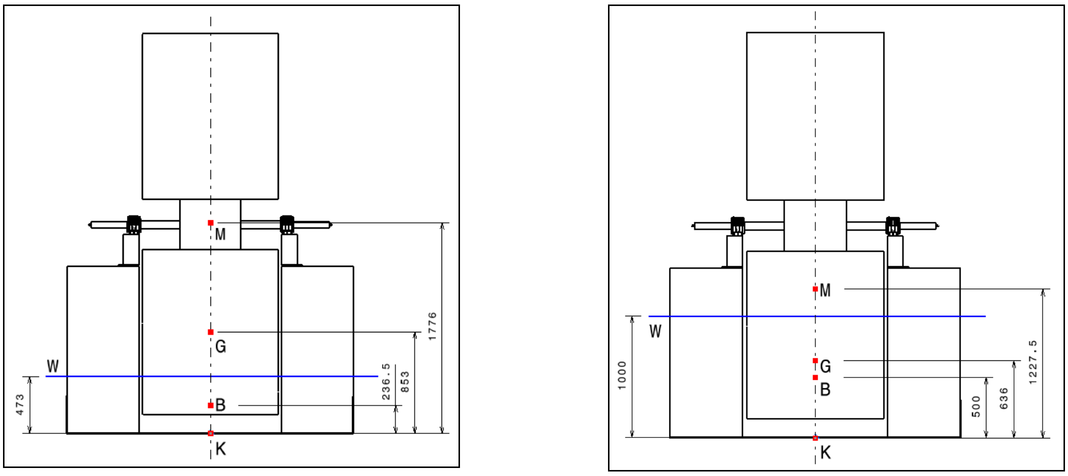
Metacentric height has also be calculated at heeling angles and angles of up to 30° (with draught of 1,000 mm) the LSM is able to stabilize itself. The metacentric height $\overline{GM}$, the distance between the centre of gravity and the metacentre, has a direct relationship with the rolling period of a floating body. The rolling period is the time (*T*) from the biggest deviation on starboard side to the biggest deviation on port side and back. The rolling period can be estimated from the Euler equation:

$$T = \frac{2\pi k}{\sqrt{g\,\overline{GM}}} \tag{3}$$

where *k* is the radius of gyration through the longitudinal axis,

$$k = \sqrt{t_{min}^2 + (B/2)^2}\,. \tag{4}$$

The rolling period of the LSM without ballast amounts 2.7 s. The other rolling periods are shown in the Table 4 below. The lower the LSM sinks into the water, the longer the rolling period.



(a) Un-blasted with draft of 473mm **(b) Ballasted with draft of 1000mm**

Figure 5 Positions of the centre or gravity, bouncy and metacentric height.

Table 3 Overview of different draughts and corresponding metacentre

Draught [mm]	Add. Ballast [kg]	Centre of Gravity (z) [mm]	Centre of Buoyancy (z) [mm]	Metacentre [mm]
473	0	853	237	1,780
750	2,520	657	375	1,350
1,000	4,800	636	500	1,220

Table 4 Overview of different rolling periods depending on draught

Draught [mm]	k [mm]	$\overline{GM}$ [mm]	T [s]
473	1,290	923	2.7
750	1,415	689	3.5
1,000	1,562	591	4.1

3.3. Initial towed results

To-date only towing tests have been performed with the large scale model. These are being performed in the naval base of Rostock on calm water. The tow runs are 200m in length and achieved be a driving an electro-hauler at constant speed. The working boat is required to setup the model in the correct position. Towing velocities from 0.5 m/s to 1.5 m/s has been achieved with this setup. The tow speeds are recorded using GPS and the power is measured as described in Section 3.1. Water levels and flow speeds inside the model are also recorded. The towing setup is shown in Figure 6.

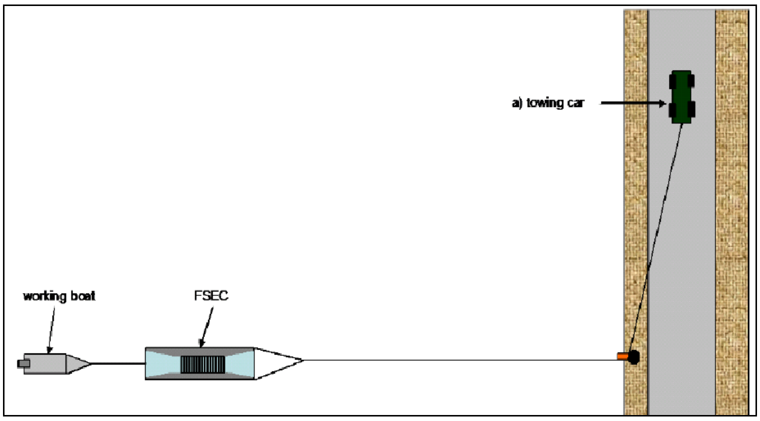


Figure 6 Towing setup

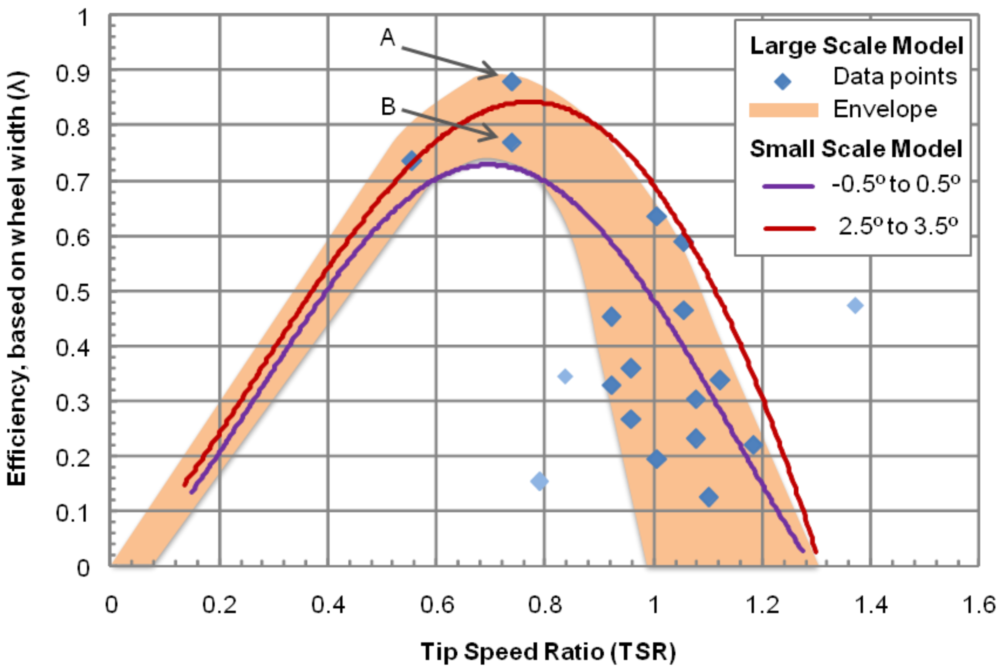


Figure 7 Typical initial efficiency results from towing tests at 1.0 and 1.4 m/s

Figure 7 show some initial results from the first phase of testing. The data in the plot has large scatter as it includes different model drafts and clearances between the wheel and base plate. These results however show similar tends to the small scale model results with peak efficiency of about 0.8 at TSR of about 0.8. Point A is at a draft of 0.65 m corresponding to a diameter to blade emersion ratio of about 5. Point B has a deeper draft of 0.75 m and an emersion ratio of about 4.2 and although the efficiency is 11% less the power is 4% higher. The power is higher as there is more kinetic energy available but the efficiency is reduced due to more turbulent flow and water lifting as the blade exits the water. These depth ratios are within the range of 4 to 5 suggested by Bresse (1876).

The towing test campaign is still ongoing and the further results should hopefully establish model scaling.

4. IMPLICATIONS AND FUTURE WORK

The small scale model tests have shown a small drop in efficiency for floating models and improvement in efficiency with the bow trimmed down. The current large scale model tests are similar in performance but further tests are required to fully understand the trends.

On completion of the towed tests the large scale model will be tested in a river Warnow, Germany to investigate the effect on the flora/fauna habitat and then in stronger tidal currents behind the Ems barrier, Germany. Further details are in Weichbrodt *et al.* (2010) and Hadler and Broekel (2010 & 2011).

Initial scaled sizes for a plausible 10kW machine are presented in Table 5 for three flow speeds. This shows the large sizes required for low flow speeds but attractive geometry if a suitable high speed site could be found, such a tidal estuary or river bend. Floating water wheels may not be a solution for large scale renewable energy production but due to their simplicity and low costs they may be a viable option for decentralised electricity generation in developing countries or remote locations.

Table 5 Scaled sizes for 10 kW machine

Power (kW) **Flow speed (m/s)**	**10.0** **1.6**	**10.0** **1.8**	**10.0** **2.0**
Wheel width (m)	2.6	2.2	1.9
Wheel diameter (m)	8.3	7.0	6.0
Overall length (m)	19.8	16.6	14.1
Overall width (m)	5.6	4.7	4.0

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